IMPINGEMENT HEAT TRANSFER FROM TURBULENT AIR JETS TO FLAT PLATES - A LITERATURE SURVEY

by John N. B. Livingood and Peter Hrycak

Lewis Research Center

Cleveland, Ohio 44135

1. Report No. NASA TM X-2778	2. Government Accession No.	3. Recipient's Catalog No.	
4. Title and Subtitle IMPINGEMENT HEAT TRANSFER FROM TURBULENT AIR JETS TO FLAT PLATES - A LITERATURE SURVEY		5. Report Date May 1973	
		6. Performing Organization Code	
7. Author(s) John N. B. Livingood, Lewis Research Center; and Peter Hrycak, Newark College of Engineering, Newark, New Jersey		8. Performing Organization Report No. E-7298	
		10. Work Unit No. 501-24	
9. Performing Organization Name and Address Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio 44135		11. Contract or Grant No.	
		13. Type of Report and Period Covered Technical Memorandum	
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546			
		14. Sponsoring Agency Code	

15. Supplementary Notes

16. Abstract

Heat-transfer characteristics of single and multiple turbulent air jets impinging on flat surfaces have been studied by many investigators. Results of many of these studies are summarized herein. Suggested correlations for use in the design of cooled turbine blades are noted, and areas where further research would be advisable are identified.

17. Key Words (Suggested by Author(s)) Literature survey Impingement cooling Heat transfer Flat plates	18. Distribution Statement Unclassified - unlimited		
19. Security Classif. (of this report) Unclassified	20. Security Classif. (of this page) Unclassified	21. No. of Pages 43	22. Price* $3.00

IMPINGEMENT HEAT TRANSFER FROM TURBULENT AIR JETS TO FLAT

PLATES - A LITERATURE SURVEY

by John N. B. Livingood and Peter Hrycak[*]

Lewis Research Center

SUMMARY

Analytical and experimental investigations of the heat-transfer characteristics of single and multiple turbulent jets of air impinging on flat surfaces are reviewed. Both circular jets and slot jets are considered, and the effect of crossflow on impingement heat transfer is included.

Comparisons of theory and experiment and of experimental results from different investigations are given when possible. Some correlations are suggested for use in turbine vane and blade design, and areas where additional research is in order are noted.

INTRODUCTION

This report presents a concise review of the available literature on the heat transfer from air jets impinging on flat surfaces. Impingement cooling began finding use in the cooling of gas turbine blades and vanes starting in the early 1960's. It has found favor as a means of cooling in regions of high heat flux because it is a more effective method of cooling than ordinary convection cooling. The use of single jets, rows of jets, arrays of jets, single slots, and rows of slots, particularly impinging on flat surfaces, has been studied by many investigators. Turbine vane and blade midchord regions are sometimes approximated as flat surfaces, and this review is aimed at determining which of the many correlations may be most applicable for future turbine vane and blade designs and the areas where further research may be warranted. Nozzle-to-surface spacing is one important parameter to be considered; and when large areas are to be impingement cooled, jet row-to-row spacing and crossflow effects must be considered.

---

[*]Professor of Mechanical Engineering, Newark College of Engineering, Newark, New Jersey.

The use of impinging air jets complicates both the flow and heat-transfer fields and, in comparison with ordinary convection cooling, relatively little information concerning either flow or heat transfer is known. As a consequence, the NASA Lewis Research Center let a contract to the Newark College of Engineering to investigate the fundamentals of impingement flow and heat transfer. References 1 and 2 cover the flow studies; the present report is a result of the literature survey made relative to the heat-transfer studies under the contract.

JET FLOW CHARACTERISTICS

A knowledge of the velocity of the jet is required in establishing heat-transfer coefficients along an impingement surface. With this in mind, it seems appropriate to reiterate herein some information relative to impinging jet flow presented in references 1 and 2.

Four distinct flow regions can be characterized for impinging turbulent jets. Figure 1(a) shows these four regions:

(1) Region I is the region of flow establishment. It extends from the nozzle exit to the apex of the potential core. The potential core is the central portion of the flow in which the velocity remains constant and equal to the nozzle exit velocity. For turbulent flow, the potential core is from 6 to 7 nozzle diameters long for a circular nozzle and from about 4.7 to 7.7 slot widths long for a slot jet.

(2) Region II is a region of established flow in the direction of the jet beyond the apex of the potential core. It is characterized by a dissipation of the centerline jet velocity and by a spreading of the jet in the transverse direction. For a circular jet, the axial velocity is inversely proportional to the distance from the jet nozzle expressed in nozzle diameters. For a slot jet, it is inversely proportional to the square root of the distance from the jet nozzle expressed in slot widths. The constants of proportionality are the potential-core length and the square root of the potential-core length for circular and slot jets, respectively.

In reality, the distances referred to should be measured from the true jet origin instead of from the nozzle. However, according to reference 3, no calculation method is as yet available for fixing the exact location of the true jet origin. Furthermore, the distinction between the true jet origin and the nozzle becomes insignificant when the plate is located several diameters away from the nozzle.

(3) Region III is that region in which the jet is deflected from the axial direction. A method for calculating the radial velocity along the impingement plate is discussed in reference 1.

(4) Region IV is the wall jet region. Several methods for calculating the velocity profiles through the wall jet region are discussed in reference 1.

When rows of holes or rows of slots are considered (and this would be the case for impingement-cooled turbine vane and blade suction and pressure surfaces), care must be exercised in obtaining proper spacing so that the jets do not interfere with each other. Information on jet spreading is reported in reference 1. In addition, when multiple jets are considered, the effects of crossflow on heat transfer must be included. In some of the work reviewed in the following sections of this report, crossflow effects were considered.

The influence of the nozzle diameter on impinging jet flow patterns (see fig. 1(b) and ref. 2) and the jet turbulence scale effect on stagnation-point heat transfer both indicate a weakening of the fluid mechanic similarity. The similarity usually attached to geometrically similar systems is reduced in some instances and the resulting heat-transfer correlations may become strictly representative only of the experimental apparatus involved in these cases.

ANALYTICAL HEAT TRANSFER

Techniques Used and Results Obtained

Experimental evidence shows that maximum heat transfer is achieved when the target plate is located at a distance from the nozzle equal to the potential-core length. When the nozzle-to-plate spacing exceeds the potential-core length, the intensity of the heat transfer decreases with the dissipation of the original jet velocity. The region of the jet impingement (region III of fig. 1) was investigated by numerous individuals who employed similarity techniques originating from free-jet studies. After the jet hits the target, it is deflected and eventually a wall jet develops (region IV of fig. 1). The wall jet region exhibits boundary-layer characteristics and may be analyzed by boundary-layer theory.

Kezios. - Initial attempts to analytically predict heat-transfer from an impinging jet to a plane surface were limited to the neighborhood of the stagnation point. In this region, even for turbulent jets, laminar flow predominates in the boundary layer, especially near the stagnation point. Reference 4 reports the following expression for the stagnation-point heat transfer:

$$Nu_o = 0.475 \sqrt{Re_{D/2}}$$

where the nozzle radius $D/2$ was used as the characteristic dimension. Away from the stagnation point, the local Nusselt number was found to yield a maximum at $r/(d/2) = 1.3$, that is,

$$Nu = F \frac{r}{d/2} \sqrt{Re_r}$$

Walz. - Walz (ref. 5), using the results originally derived by Eckert, obtained the expression

$$Nu = 0.44 \, Pr^{0.36} \, Re_r^{0.5}$$

for the local values in the axisymmetrical flow, and

$$\overline{Nu} = 0.88 \, Pr^{0.36} \, Re_r^{0.5}$$

for the average values in the axisymmetrical flow.

Metzger. - For slot jets, using Eckert's wedge flow solution, Metzger (ref. 6) obtained the equation

$$Nu = 0.57 \, Pr^{0.37} \, Re_r^{0.5}$$

for local values of the Nusselt number based on distance from the stagnation point and

$$\overline{Nu} = 1.14 \, Pr^{0.37} \, Re_r^{0.5}$$

for the average value of the Nusselt number.

The inability of such formulas to account directly for the effect of the nozzle-to-plate separation distance must be considered their principal weakness.

Schauer. - Reference 7 presents the solution for the hydrodynamics, the basis for the analysis in reference 8. In reference 8, the thermal-boundary-layer equations for incompressible turbulent flow were solved by applying integral techniques. A modified Reynolds analogy (relating the diffusivity for heat to the diffusivity for momentum) was used to link the thermodynamic to the hydrodynamic solution.

Cadek. - In the impingement region, Cadek (ref. 9) used an approach that, although based on the integral techniques, was reducible to Eckert's wedge flow solution. In the wall jet region, Cadek's efforts resulted in complicated expressions which reduced to the well-known flat-plate solution for the corresponding limiting case.

Cadek stressed that in all analytical solutions in the impingement region one should keep in mind one important fact: In the wedge flow class of solutions an infinite uniform stream impinges on a surface, whereas in our present problem we deal with a nonuniform jet of finite width striking a surface. In the first case, the Euler number m is constant; in the latter case, it is a variable function of distance from the stagnation point.

4

Eckert. - Eckert and Livingood developed methods of calculating heat transfer with a variable m in reference 10. According to Cadek, his method gives results on heat-transfer rates that were essentially similar to those obtainable by Eckert and Livingood's approach while at the same time yielding more information on the boundary layer itself.

Tomich. - Tomich also calculated heat transfer from impinging, turbulent jets (ref. 11). To eliminate the need for various simplifying assumptions, Tomich took the approach of solving the governing equations by means of finite difference techniques. Solutions were extended into the compressible flow region, and it was found that the jet Mach number and jet temperature ratio were the only two initial jet properties necessary to characterize the dimensionless velocity and temperature variations in the free jet.

Brdlik. - Another analytical approach has been taken in reference 12 by Brdlik and Savin. They used the integrated energy equation adapted for situations with a radial symmetry and obtained, for $z_n/D < 6.2$, based on the plate radius $d/2$

$$\overline{Nu} = 1.09 \, Pr^{1/3} \, Re_{d/2}^{1/2}$$

which shows reasonably good agreement with the test data of several investigators.

Discussion of Analytical Solutions

In discussing the available analytical solutions, it is seen that they have certain common features. The general form of the equation for the average Nusselt number near the stagnation point is

$$\overline{Nu} = C \, Pr^m \, Re^n$$

with $m = 1/3$ and $n = 1/2$.

The value of C varies from 0.88 (ref. 4) to 1.09 (ref. 12) for circular jets and is 1.14 for slot jets (ref. 6), according to Eckert's method. For the wall jet region, good agreement between theory and experiment was found by several investigators, as is pointed out later in this survey.

EXPERIMENTAL HEAT TRANSFER

Single Circular Jets

Perry. - Experimental results obtained from the study of heated jets striking cool flat surfaces are reported in references 13 to 15. Perry (ref. 13) considered air heated

to 873 K $(1112^{\circ}$ F) discharging through nozzles 1.65 and 2.16 centimeters (0.65 and 0.85 in.) in diameter at velocities to 91 meters per second (300 ft/sec). The nozzle-to-plate spacing was at least 8 nozzle diameters. The effect of the angle of jet impingement was studied. The heat flow was determined by means of a 1.65-centimeter (0.65-in.) diameter calorimeter; this dimension, d', was used as the characteristic dimension in both the Nusselt and Reynolds numbers.

The test results are shown in figure 2 and were correlated by

$$Nu = K \, Re_{d'}^{0.7} \, Pr^{0.33}$$

where the value of K is dependent on the angle of impingement as follows:

Angle of impingement, deg	$K = Nu \, Re_{d'}^{-0.7} \, Pr^{-0.33}$
90	0.1810
75	.1745
60	.1579
45	.1422
30	.1224
15	.1037

Perry found about a twofold increase in the plate heat transfer as the jet axis of symmetry was turned from 15° to 90° with respect to the plate.

Thurlow. - The influence of nozzle-to-plate spacing on the heat transfer of an impinging hot air jet on a flat plate was investigated by Thurlow (ref. 14). Values of $z_n/D > 10$ and Re_D to 60 000 were considered. Nozzles 2.54 and 1.27 centimeters (1 and 1/2 in.) in diameter were used to impinge hot air on a copper plate 61.0 by 15.2 centimeters (24 by 6 in.). The data were correlated by

$$\overline{Nu} = C \, Re_D^{1/3} \, \exp\left(-0.037 \, \frac{z_n}{D}\right)$$

where the coefficient C was found to be equal to 1.06 for the 2.54-centimeter (1-in.) diameter nozzle and equal to 0.33 for the 1.27-centimeter (1/2-in.) diameter nozzle. In other words,

6

$$C \sim D^{3/2}$$

with D expressed in inches.

Huang. - Huang (ref. 15) investigated the heat transfer from a hot turbulent air jet to a plate normal to the jet flow. Nozzle diameters of 0.32, 0.40, 0.48, 0.56, and 0.64 centimeter (1/8, 5/32, 3/16, 7/32, and 1/4 in.); nozzle-to-plate spacings from 1 to 12 diameters; and nozzle exit Reynolds numbers to 10^4 were investigated. Both stagnation and average data were correlated according to

$$\text{Stagnation:} \quad Nu_o = 0.0233 \, Re_a^{0.87} \, Pr^{0.33}$$

$$\text{Average:} \quad \overline{Nu} = 0.018 \, Re_a^{0.87} \, Pr^{0.33}$$

Here, the Reynolds number is based on the impact velocity. The impact velocity was obtained from

$$U_{imp} = \sqrt{\frac{2(p_t - p_s)}{\rho}}$$

where p_t was the total or stagnation pressure, p_s was the static pressure, and both were measured values. The experimental results showed that the maximum heat transfer was achieved with a 0.476-centimeter (3/16-in.) diameter nozzle and an r/D ratio less than 3. It should be pointed out that, in this investigation, no change was found in the heat-transfer coefficient for $z_n/D < 6$; in contrast, major changes were found in reference 16 (to be discussed next).

Gardon. - Room-temperature jets impinging on heated flat surfaces are reported in references 16 to 19. Reference 16 presents results of tests of cool air impinging from a circular jet onto an isothermal hot surface. Nozzles with diameters from 0.23 to 0.90 centimeter (0.089 to 0.354 in.) and nozzle exit velocities to sonic were used in conjunction with a 15.24-centimeter (6-in.) square aluminum plate. Heat-flow transducers 0.9 millimeter (0.035 in.) in diameter were mounted flush in the aluminum plate to measure heat-transfer rates. In a later publication, Gardon stated that calibration of the transducers could have been as much as 40 percent too high. Provisions were made to investigate various nozzle-to-plate spacings (0.25 to 40 diam).

Stagnation-point heat transfer was presented in reference 16 by basing the Nusselt number on the nozzle diameter and the Reynolds number on the nozzle diameter and nozzle exit velocity. The results are shown in figure 3 for various nozzle diameters and nozzle-to-plate spacings. The figure shows peak heat-transfer rates at progressively lower nozzle-to-plate spacings, the peaks increasing with increasing nozzle diameter.

Figure 3 shows clearly the residual effects of nozzle diameter on heat transfer for all Reynolds numbers; this effect is most pronounced for $z_n/D < 10$. For values of $z_n/D > 20$ and nozzle exit Reynolds number $> 14\ 000$, the following correlation was obtained:

$$Nu_0 = 13 \sqrt{Re_D} \frac{D}{z_n}$$

The variation of heat-transfer rates with radial distance from the stagnation point is shown in figure 4 for a nozzle diameter of 0.634 centimeter (0.250 in.), a nozzle exit Reynolds number of 28 000, and the overall temperature difference between the hot plate and the incoming impingement air of 20 K (36° F). The figure shows a bell-shaped variation for large nozzle-to-plate spacings. As the nozzle-to-plate spacing is reduced, peaks appear in the heat-transfer rates. At $z_n/D = 6$, an annular hump begins to develop and grows into a well-defined secondary peak as z_n/D decreases. When the nozzle-to-plate spacing is less than the potential-core length, the central peak no longer appears, and in its place is a central minimum of heat transfer. For values of $z_n/D < 1/2$, the impingement of a free jet gives way to a wall jet and the heat-transfer rates rise, as shown by the upper curve of figure 4. Figure 5 shows the radial variation of the normalized heat-transfer coefficients between a plate and an impinging jet.

Walz. - The cooling of flat circular plates by air jets impinging normally on the plates was investigated in reference 5. The transient technique was used to determine average heat-transfer coefficients for nozzle Reynolds numbers to 31 000, nozzle-to-plate spacings from 1 to 8 nozzle diameters, nozzle diameters from 0.32 to 0.76 centimeter (0.125 to 0.3 in.), and target diameters from 1.27 to 2.54 centimeters (0.5 to 1 in.). Results similar to the original results of reference 16 were obtained (before announcement of the possibility of inaccurate calibration of the transducers in ref. 16). The effect of the ratio of target diameter to nozzle diameter is also discussed.

Brdlik. - Experimental heat-transfer coefficients were presented in reference 12 for a jet impinging on a heated plate. The data showed that for $z_n/D \leq 6.2$, the heat-transfer coefficient was practically independent of the distance from the jet exit; and for $z_n/D > 6.2$, the distance z_n had an appreciable effect on the heat transfer. Good agreement was obtained when the experimental and analytical results were compared.

Chamberlain. - Room-temperature turbulent air jets impinging on a segmented flat Invar steam-heated plate perpendicular to the jet axis were studied by Chamberlain (ref. 17). Two nozzle sizes, 0.634 and 0.952 centimeter (0.250 and 0.375 in.) in diameter; nozzle exit Reynolds numbers to 67 000; and nozzle-to-plate spacings to 50 nozzle diameters were investigated. When the value of nozzle-to-plate spacing z_n/D was 7 or less (the plate was located within the potential-core region), local stagnation-point Nusselt numbers Nu_0 were correlated by

$$Nu_o = 1.16 \, Re_D^{0.447} \, Pr^{0.333} \qquad \text{for} \quad z_n/D \leq 7$$

When the plate was located outside the potential-core region, the data were correlated by

$$Nu_o = 0.384 \, Re_a^{0.569} \, Pr^{0.333} \qquad \text{for} \quad z_n/D > 7$$

where Re_a is the Reynolds number based on the arrival velocity (as proposed in ref. 16 where multiple-jet data were correlated). Results of thermal conductivity tests conducted by the National Bureau of Standards on a sample of Invar used in the test plate were 40 percent higher than those previously available. The coefficients in the two preceding equations have been modified accordingly.

For locations along the plate away from the stagnation point, the correlation

$$\frac{h_r}{h_o} = \exp\left[-1.56\left(\frac{r}{z_n}\right)^{0.75}\right]$$

was suggested; here h_r is the local coefficient at a distance r from the stagnation point.

Donaldson. - Heat-transfer coefficients for a jet impinging on a plate are reported in reference 18 for jet exit velocities of 61 to 213 meters per second (200 to 700 ft/sec). Thin calorimeters were used to obtain the heat-transfer distributions on the plate. Turbulence characteristics of the same free jets were measured and used as a basis for correction factors to be applied to computed laminar stagnation-point heating. From the results, the reference concludes that the following procedure should yield results that will suffice for most engineering purposes:

(1) Near the stagnation point, the heat transfer should be computed from the laminar heat transfer that would take place on a surface having the same pressure distribution as that on the impingement plate and using a correction factor obtained from figure 6; note the abscissa of figure 6 is z_n/D and not the Reynolds number.

(2) Farther away from the stagnation point, the heat transfer should be computed on the basis of the technique of reference 19.

(3) Very far from the stagnation point, the relation

$$Nu = 0.12 \, Re_{y_{1/2}}^{0.8}$$

should be used, where the Reynolds number is based on the distance at which the velocity is one-half the maximum free-jet velocity in the plane of impingement.

Arrays of Circular Jets

Freidman. - Reference 20 presents results of an experimental investigation of heat transfer for an array of circular air jets impinging on a heated plate. Effects of hole size and spacing, distance between the array of nozzles and the heated plate, and air velocity on the heat-transfer rates were studied. Hole sizes from 0.634 to 1.905 centimeters (0.250 to 0.750 in.) and spacings between the nozzles and the heated plate from 5.72 to 15.88 centimeters (2.25 to 6.25 in.) were considered. Results indicated that spacings between holes of from 4 to 6 hole diameters (giving about 5 to 2 percent free-flow area) yielded the best heat-transfer results. Attempts were made to compare the results with those of other investigators. These attempts were unsuccessful because of an insufficient range of experimental data and because of the complexity of the flow patterns.

Gardon. - In addition to the study of single-jet impingement already discussed, Gardon also considered multiple arrays of jets; the results are reported in reference 16. In this study, nozzles with diameters from 0.159 to 1.27 centimeters (1/16 to 1/2 in.) were arranged in square arrays. A 5 × 5 nozzle array on 5.08-centimeter (2-in.) centers and a 7 × 7 nozzle array on 3.05-centimeter (1.2-in.) centers were used. Airflow rates from 244 to 3906 $kg/(hr)(m^2)$ (50 to 800 $lb/(hr)(ft^2)$) were considered. (Nozzle exit velocities were low enough that compressibility effects were negligible.)

When attempts to correlate the data based on nozzle exit velocity failed, the authors of reference 16 decided to correlate by using the arrival velocity. Space-averaged heat-transfer data were successfully correlated when the center-to-center hole spacing was used as the characteristic dimension in the Nusselt and Reynolds numbers and the arrival velocity was used in the Reynolds number. The arrival velocity U_a for nozzle-to-plate spacings greater than 8 was evaluated as

$$U_a = 6.63 \, U \frac{D}{z_n} \qquad \text{for} \quad z_n/D > 8$$

so that

$$Re_a = 8.45 \frac{G' x_n^3}{z_n D \mu} \qquad \text{for} \quad z_n/D > 8$$

where G' is the mass flow rate of air per unit area of nozzle array. The average Nusselt number was found to be

$$\overline{Nu} = 0.286 \ Re_a^{0.625}$$

Although the equation for Re_a is valid only for $z_n/D > 8$, the equation for $\overline{Nu}$ holds down to $z_n/D > 1$.

The effect of nozzle-to-plate separation distance on the local Nusselt number was also investigated. It was found that the shapes of the curves, notably different, depend on Re_a as well as on z_n/x_n and, when $z_n/D < 8$, on z_n/D also. When $z_n/D > 8$, it was found that the normalized amplitudes $(h_{max} - h_{min})/\overline{h}$ were inversely proportional to $(z_n/x_n)^{0.8}$ and practically independent of Re_a between 25 000 and 250 000.

Huang. - Since in reference 15 the maximum heat transfer for a single jet was found for a 0.476-centimeter (3/16-in.) diameter nozzle and an $r/D < 3$, the investigation of multiple jets was limited to the 0.476-centimeter (3/16-in.) diameter nozzle, spaced at 1.588 centimeters (5/8 in.) center-to-center spacing in a single row and the row-to-row distances Y of 3.81, 7.62, and 15.24 centimeters $\left(1\frac{1}{2}, 3, \text{ and } 6 \text{ in.}\right)$. The centerline and the average heat-transfer coefficients between rows obtained are shown in figure 7 as a function of nozzle-to-plate spacing. When $z_n/D > 20$, all the curves of figure 7 converge and the heat-transfer coefficient becomes equal for all values of Y/D. For values of $z_n/D < 20$, the centerline heat-transfer coefficient increases with increasing values of Y/D; and the average coefficient between rows of holes decreases with increasing values of Y/D.

Ott. - A triangular array of circular jets impinging on a flat surface was investigated in reference 21. Only a limited range of Reynolds numbers was considered and only z_n/D and Re were variable. An average value of the heat-transfer coefficient was therefore correlated with only Re and z_n/D as parameters. No variation in free-flow area was included in this study.

Hilgeroth. - Hilgeroth (ref. 22) discovered large deviations in heat transfer caused by jet interaction and spent air disturbances. He found that, for x_n/D, z_n/D, and U held constant, the heat-transfer coefficients increased as the hole diameter increased. A 25 percent decrease in heat-transfer coefficient between z_n/D of 2 and 6 was observed. Maximum values of h were obtained for z_n/D of 6.3 for a free-flow area of 1.5 percent. The Reynolds number exponent was found to be a function of hole-spacing-to-hole-diameter ratio, increasing as x_n/D increased from 3.5 to 12.5. A square array of jets was found to be superior to an equilateral triangular array for the same jet velocity and open-area ratio.

Kercher. - References 23 and 24 reported the investigation of a series of configurations

(1) To measure the effect of variable hole diameter at constant hole spacing

(2) To measure the effect of variable hole spacing at constant hole diameter

(3) To test the ability to scale configurations at constant open-area ratios

(4) To test wide variations in the ratio of crossflow to impingement flow

(5) To test wide variations in open-area ratio

Figure 8(a) shows the effect of varying hole diameter (or increasing the open-area ratio) at constant hole spacing. The figure shows good agreement between the data obtained by Kercher and those obtained by Huang (ref. 15), Ott (ref. 21), and Freidman and Mueller (ref. 20).

Figure 8(b) shows the effect of varying the hole spacing (or open area) for constant hole diameter. Better heat transfer is achieved by increasing the number of jets (decreasing the hole spacing). The discrepancies between Kercher's data and the results of the study of Gardon and Cobonpue (ref. 16) are attributed to the influence on boundary-layer phenomenon after jet impingement from perforated plates and from long throat nozzles. Satisfactory agreement between the data of Kercher (refs. 23 and 24) and Hilgeroth (ref. 22) is shown in figure 8(c).

Data indicated that the average surface heat-transfer coefficients increased as z_n/D was increased from 1 to about 5 (see ref. 23). If Re_D is considered, the exponent of Re_D was found to increase with increasing z_n/D, a result also found for a single jet by Huang (ref. 15).

In reference 23, a correlation equation was developed which depended on determining three parameters by graphical means. One of these parameters involves the effects of crossflow on heat transfer.

The test results are summarized in reference 24 as follows: (1) heat transfer by a multiple square array of round impinging jets on a flat plate cannot be correlated by power-function expressions of dimensionless parameters; (2) heat-transfer coefficients increase with increasing open area A_f; (3) heat transfer is dominated by the hole-diameter Reynolds number Re_D and the hole-spacing-to-hole-diameter ratio; (4) the exponent on Reynolds number is a strong function of x_n/D with a distinct change in value at Reynolds number Re_D near 3000; (5) within the range tested, increasing z_n/D increases heat transfer without crossflow but decreases heat transfer with crossflow; (6) increasing crossflow to the jet decreases heat-transfer performance; (7) decreasing hole diameter with increasing number of holes, everything else being equal, improves heat-transfer performance; and (8) the correlation proposed shows good agreement with the measured results.

Burggraf. - Although reference 25 deals primarily with impingement of a row of circular jets into a half-cylinder, impingement of a row of four circular jets of 0.653-centimeter (0.257-in.) diameter and 2.164-centimeter (0.852-in.) center-to-center spacing on a flat plate was also reported. Both plate and cylinder data were correlated by basing average Nusselt and Reynolds numbers on plate width.

Metzger. - An experimental study of the effects of crossflow on the heat-transfer characteristics of single rows of air jets impinging on plane surfaces is reported in reference 26. A row of 10 circular jets, each 0.254 centimeter (0.1 in.) in diameter with

12

center-to-center spacing ratios x_n/D from 2.5 to 5 were considered. Nozzle-to-plate spacings z_n/D from 2 to 6.7 and ratios of crossflow to jet flow M^* from 1 to 3 were studied. From previously available impingement data and on the basis of geometric similarity, the average Stanton number should be able to be correlated by

$$\overline{St} = St\left(Re_{es}, \frac{x_n}{D}, \frac{z_n}{D}, \frac{l}{D}\right)$$

where $\overline{St}$ is $\overline{h}/(Gc_p)$ evaluated over the cooled surface of half-length l and Re_{es} is based on the width of a slot whose area is equal to that of the 10 holes used, that is,

$$Re_{es} = \frac{2GB_{es}}{\mu}$$

An extra parameter $M^* = m_c/m_j$ should correlate data with crossflow (see fig. 9). Negative values of r denote distances upstream from the nozzle exit plane, and positive values of r denote distances downstream. Test results were plotted as $\overline{St}$ (with crossflow) divided by $\overline{St}_{M^*=0}$ (without crossflow) as the ordinate.

Crossflow effects on heat transfer for a given value of equivalent-slot Reynolds number ($Re_{es} = 2000$) and a given hole spacing ($x_n/D = 3$) because of nozzle-to-plate spacing variations are shown on figure 9. The figure shows that crossflow destroys the symmetry about the nozzle exit plane by deflecting the jets downstream, with a resulting increase in downstream heat transfer and a decrease in upstream heat transfer. Overall performance over the region $r/D = \pm 10$ about the nozzle exit plane showed in all cases that the average heat transfer was less than that for the case of no crossflow.

Figure 10 shows the effects of hole spacing on crossflow heat transfer for a nozzle-to-plate spacing of 2 nozzle diameters. The figure shows the importance of hole spacing; and, as the hole spacing increases, the ratio $\overline{St}/\overline{St}_{M^*=0}$ approaches unity both upstream and downstream of the nozzle exit plane. As M^* increases, the general trend shows a decrease in $\overline{St}/\overline{St}_{M^*=0}$ in the region upstream from the nozzle exit plane and an increase in the downstream region.

For the ranges of variables covered in this investigation, the crossflow heat-transfer results were correlated by

$$\overline{St} = 0.0822\, M^{*-0.049}\, Re_D^{-0.338}$$

This correlation, and all the data, are shown in figure 11.

Single Slot Jets

Metzger. - An experimental study of the heat-transfer characteristics of slot jets impinging normally on a flat target was performed by Metzger (ref. 6). Nozzles with lengths of 1.90 centimeters (0.75 in.) and widths of 0.025, 0.051, 0.102, 0.15, and 0.204 centimeter (0.01, 0.02, 0.04, 0.06, and 0.08 in.) were studied as they impinged on two copper targets of half-lengths 1.27 and 0.635 centimeter (0.50 and 0.25 in.) mounted in blocks of balsa wood. Reynolds numbers to 10 000 (based on nozzle exit velocity and nozzle hydraulic diameter) were considered.

Initial tests, with all variables except nozzle-to-target spacing held constant, showed that maximum average heat transfer occurred at a value of $z_n/B_0 = 8$. Since no significant variation was noted as z_n/B_0 was varied from 7 to 10, it was decided that all other tests would be made for $z_n/B_0 = 8$. This value of 8 compares with the value of z_n/D of between 6 and 7 for a single circular jet.

Average and local heat-transfer data were obtained. The average data were measured, and local data were obtained from the average by

$$St(r) = \overline{St}(r) + r\left[\frac{d\ \overline{St}(r)}{dr}\right]$$

The exponent on the nozzle exit Reynolds number was found to be 0.434 from plotting $\overline{St}$ against Re_d. The correlation equation was found to be

$$\overline{St}\left(Re_d\right)_h^{0.434} Pr^{0.63} = 0.74\left(\frac{d}{B_0}\right)^{-0.434}$$

for $7 < z_n/B_0 < 10$ and $3 < d/B_0 < 50$.

In order to gain an insight into the relative performance of slot jets and circular jets, one circular nozzle was also tested by Metzger. He investigated nozzle-to-target spacings of 1, 5, and 10 nozzle diameters and did not find any maximum heat transfer in this range. He did conclude that the shape of the area to be cooled is of primary importance in deciding between the use of a slot jet or a circular jet; however, this conclusion was reached by consideration of $z_n/D = 1$ and not at the value for a maximum as found by other investigators.

Myers. - Experimental wall jet heat-transfer data were obtained by using a 1.27-centimeter (0.5-in.) wide slot and slot Reynolds numbers from 16 600 to 38 100 and are reported in reference 8. Data were taken for values of r/B_0 from 20 to 186 for five different unheated starting lengths L (r is the distance along the wall). The results are shown in figure 12; the solid line is the analytical solution and the dashed line is the best

correlation of the data for $r/B_o > 45$, namely,

$$\text{St } Re_{B_o}^{0.2}(10^2)\left[1 - \left(\frac{L}{r}\right)^{9/20}\right]^{1/16} = 12\left(\frac{r}{B_o}\right)^{-0.5658}$$

For design purposes, it was recommended that the right member of the preceding equation be replaced by $11.8(r/B_o)^{-9/16}$; the exponent 9/16 is very close to the analytically determined exponent 9/15, and the error in changing the right member as suggested is less than ±0.5 percent.

Gardon. - Heat-transfer characteristics of air jets issuing from slots 0.159, 0.317, and 0.635 centimeter (1/16, 1/8, and 1/4 in.) wide and 15.24 centimeters (6 in.) long and impinging normally on an aluminum 15.24-centimeter (6-in.) square plate are reported in reference 27. Both average and local heat-transfer rates were obtained, the latter by use of a 0.9-millimeter (0.035-in.) diameter transducer. Nozzle-to-plate spacing was also varied. Maximum stagnation heat transfer was found to be a function of the nozzle exit Reynolds number and occurred between values of z_n/B_o from 7 to 10. For turbulent jets (Re > 2000) and nozzle-to-plate spacings greater than 14 slot widths, the stagnation-point heat-transfer coefficients were correlated within ±5 percent by

$$Nu_o = 1.2 \, Re_{B_o}^{0.58}\left(\frac{z_n}{B_o}\right)^{-0.62}$$

for Re_{B_o} to 50 000 and z_n/B_o to 60. Nozzle size was found to influence the turbulence levels of the jets; for Re_{B_o} = 11 000, the initial turbulence levels were found to be 0.6, 2.5, and 7.5 percent for the 0.159-, 0.317-, and 0.635-centimeter (1/16-, 1/8-, and 1/4-in.) slots, respectively.

Variations of the local heat-transfer coefficients along the plate are best described by the following, taken from reference 27:

(1) For nozzle-to-plate spacings z_n/B_o greater than 14 slot widths, the variation of heat-transfer coefficients has a bell shape.

(2) For $14 > z_n/B_o > 8$, the bell shape is modified slightly by an abrupt change in slope in the vicinity of $r/B_o = 4$.

(3) As z_n/B_o is reduced below 8, two "humps" begin to form at about $r/B_o = ±7$; and for $z_n/B_o < 6$, they become well-defined secondary peaks in the heat-transfer rate. These secondary peaks are ascribed to a transition from laminar to turbulent boundary layers.

(4) As z_n/B_o is reduced below 1/2, the impinging jet becomes a "wall jet" and heat-transfer coefficients rise sharply with increasing velocities in the gap between the

nozzle exit and the plate. In this regime, the variation of local heat-transfer coefficients has two peaks on either side of the centerline.

The variation of local heat-transfer coefficients was obtained only for fully developed jets with their characteristic bell-shaped distributions. The corresponding normalized local heat-transfer coefficients h/h_o depend on the Reynolds number and the dimensionless distance from the stagnation point. When the latter is expressed in terms of r/z_n, rather than as r/B_o, nozzle-to-plate spacing z_n/B_o and slot width B_o do not directly enter into this correlation. Though developed primarily for $z_n/B_o > 14$, it holds, for all practical purposes, down to $z_n/B_o = 8$.

Average heat-transfer coefficients for the three slot widths considered over two target widths are correlated against the Reynolds number based on the arrival velocity in figure 13. The data were correlated by

$$\overline{Nu} = 0.36 \, Re_a^{0.62}$$

for $Re_a > 10^4$, $z_n/B_o > 8$, and $d/B_o > 6$. Superimposed on the figure are some data taken from reference 6, showing excellent agreement. In comparison, single round jets have average heat-transfer coefficients which correlate by

$$\overline{Nu} = 0.78 \, Re_a^{0.55}$$

for $Re_a > 10^4$, $z_n/D > 12$, and $1 < d/D < 24$. This latter correlation is applicable to new data obtained for this comparison and to round-jet data obtained by both Metzger (ref. 6) and Perry (ref. 13). Reference 27 is the publication previously referred to in which Gardon stated that the calibration of the transducers of reference 16 may have been 40 percent high.

Cadek. - The heat-transfer characteristics for a slot jet impinging on a flat copper surface for slot widths of 0.625, 1.25, and 1.875 centimeters (1/4, 1/2, and 3/4 in.), nozzle-to-plate spacings from 2 to 16 slot widths, and slot Reynolds numbers from 4600 to 102 000 were studied by Cadek (ref. 9). Local heat-transfer rates were measured by use of a miniature circular foil heat-flow sensor with a 0.091-centimeter (0.036-in.) diameter sensing area. Values of r/B_o from 0 to 36 were considered.

Local heat-transfer data are presented in figure 14 for one nozzle slot width, one Reynolds number, and several nozzle-to-plate spacings. Also shown are data from reference 27. Maximum heat transfer was found for $z_n/B_o = 8$ in both investigations, as the figure clearly shows.

Average heat-transfer data are compared with Metzger's empirical correlation in figure 15 and with Gardon and Akfirat's correlation in figure 16. It should be noted that Metzger's correlation was obtained for a Reynolds number range of 2980 to 7400 only.

16

Cadek concluded from his study that for $2 \leq z_n/B_o \leq 4$, good agreement between theory and experiment was obtained for both the stagnation region and the wall jet region. For $z_n/B_o > 8$, good agreement still was found in the wall jet region; but in the stagnation region, the measured values exceeded the theoretical ones. For ease and simplicity in design calculations, according to figure 16, use of the Gardon-Akfirat correlation for average Nusselt number is recommended.

Cartwright. - Reference 28 presents experimental heat-transfer data obtained from an impinging jet for nozzle-to-plate spacings from 8 to 47 slot widths, nozzle exit Reynolds numbers from 25 000 to 110 000, and distances along the wall jet from 0 to 132 slot widths. It was found that, at low slot Reynolds numbers, the maximum value of the heat-transfer coefficient occurred at the stagnation point and then the heat-transfer rate decreased monotonically with increasing distance from the stagnation point. At larger Reynolds numbers, the maximum heat-transfer coefficient occurred some distance from the stagnation point as a peak in the curve, quite in line with axisymmetric impinging jet heat-transfer results.

Analytical attempts to predict the heat transfer appeared adequate only in the wall jet region beyond $r/B_o = 36$. Failure of predictions in the impingement region was blamed on the high level of turbulence that exists in the free jet before impingement.

Andreyev. - An analytical method for predicting the local Nusselt number in the stagnation region of a slot jet was developed in reference 29. When discrepancies between predicted and experimental values were found, the prediction was modified, by use of some experimental data; and an empirical correlation resulted. The modification was made to account for the effects of jet turbulence. The resulting formulae were obtained:

For $1 \leq z_n/B_o \leq 6.5$:

$$Nu = 0.48 \frac{Re_{B_o}^{0.5}}{(z_n/B_o)^{0.1}} \left[1 - 0.116 \frac{(r/B_o)^2}{(z_n/B_o)^2} \right] (1 + 0.015 \epsilon)$$

For $6.5 \leq z_n/B_o \leq 12$:

$$\bullet \; Nu = 1.25 \frac{Re_{B_o}^{0.5}}{(z_n/B_o)^{0.6}} \left[1 - 1.05 \frac{(r/B_o)^2}{(z_n/B_o)^{1.4}} \right] (1 + 0.019 \epsilon)$$

For $z_n/B_o > 12$:

$$Nu = 1.25 \frac{Re_{B_o}^{0.5}}{(z_n/B_o)^{0.6}} \left[1 - 1.05 \frac{(r/B_o)^2}{(z_n/B_o)^{1.4}} \right] (1 + 0.025 \, \epsilon)$$

Array of Slot Jets

<u>Freidman</u>. - Included in the tests reported in reference 20 were plates with a multitude of slots. It was found that the slotted plates were comparable in performance to the nozzle plates except when the slotted plate forced the air out at a 45° angle with the horizontal.

<u>Gardon</u>. - Reference 27 also reports the investigation of impingement from an array of slots. Data were obtained for two combinations of slots:

(1) Three slots at 5.08-centimeter (2-in.) centers

(2) Two slots at 10.16-centimeter (4-in.) centers

It was found that, at the smallest slot-to-plate spacing, the identity of each jet was preserved and the peak heat-transfer rates of the jets differed only slightly from those corresponding to single jets. Interaction between the jets produced secondary peaks midway between the points of impingement. As the length of the jets was increased, interaction occurred before impingement, peak heat-transfer rates were no longer equal, and the peaks did not necessarily lie below the centers of the jets. At the largest spacing, the jets lost their identity and behaved almost like a large single jet.

The data were correlated on the basis of an arrival Reynolds number

$$Re_a = 2.65 \frac{Gx_n^2}{\mu \sqrt{B_o z_n}} \qquad \text{for } z_n/B_o > 7$$

with

$$\overline{Nu} = \frac{\overline{h}x_n}{k}$$

The correlation is shown in figure 17. Since the arrival Reynolds number does not consider jet interaction, it is recommended in reference 27 that the correlation

$$\overline{Nu} = 0.36 \, Re_a^{0.62}$$

should not be extrapolated to $x_n/B_o < 16$.

18

<u>Schuh</u>. - Heat-transfer data are reported in reference 30 for impingement of air through arrays of slots. Slot widths of 0.1 and 0.5 centimeter (0.394 and 1.97 in.); jet spacings of 2.5, 4, 6, and 10 centimeters (0.985, 1.576, 2.364, and 3.94 in.); and slot-to-plate spacings of 2, 4, 6, and 8 slot widths were investigated. Slot exit velocities from 4 to 100 meters per second (13.12 to 328 ft/sec) were considered. For $1200 \leq Re \leq 100\ 000$, $5 \leq x_n/B_o \leq 100$, and $z_n/B_o \approx 4$, it was found that the data could be correlated by

$$\overline{St} = 0.461\left(\frac{x_n}{B_o}\right)^{-0.327} Re_{B_o}^{-0.402}$$

within ±10 percent.

Figure 18 shows a comparison of the arrays of slot jet results of reference 30 with those of single slot jets of reference 6, for $z_n/B_o = 8$. The figure shows good agreement for the larger values of x_n/B_o of 100 and 50. For the lower values of x_n/B_o, the differences between the results for the arrays of slots and the single slot indicate the detrimental effect of the interference of the jets for closer spacings.

Schuh, in reference 30, also investigated rows of hole jets and found that lower heat transfer was achieved for the holes than for the arrays of slots at large row spacings. Little difference was found at small row spacings.

It was also found that a superimposed wall-parallel flow up to 60 percent of the jet flow could be imposed with no reduction in heat transfer. For similar conditions, but with rows of holes instead of rows of slots, some reduction in heat transfer accompanied the 60 percent superimposed wall parallel flow.

GENERAL DISCUSSION

Analytical Heat Transfer

Near the stagnation point in the stagnation region, laminar flow exists for a certain distance along the plate. Until the jet reaches a point 1 or 2 nozzle diameters from the plate, the jet behaves like a free jet. Hence, by combining free-jet potential theory with laminar-boundary-layer techniques, attempts were made to predict stagnation-region heat transfer. These attempts have been only partially successful. Inability to predict the turbulence effects accurately, the fact that the jet is not an infinite free jet, and some doubt as to the extent of the laminar boundary layer make the predictions at best approximate. Measured values were found to exceed predicted values, sometimes by a considerable amount, both for circular jets and slot jets.

Wall jet heat-transfer predictions, although so far limited to slot jets, have been found to agree with experimental results. The theoretical approaches employ the consideration of two flow layers: an inner layer along the wall and an outer layer assumed to behave as a free jet. Both integral and finite difference solutions have been obtained by different investigators, based on numerous assumptions. These solutions are complicated, and certain empirical calculations which will be recommended later are probably good enough for use in preliminary design work.

Experimental Heat Transfer

Single circular jets. - A comparison between results of heated jets impinging on room-temperature surfaces and room-temperature jets impinging on heated surfaces is difficult because of fundamentally different boundary conditions. In addition, heat-transfer rates were measured in different ways by the individual experimenters. The various methods included the use of metal rod calorimeters, steam calorimeters, thermocouples, and heat-transfer gages to measure heat-transfer rates. The areas over which these measurements were made varied in size, and what some investigators called local values were in reality at best average values. Since the correlations proposed by the individual experimenters did not specify the exact areas involved and since the characteristic dimensions used in the various correlations differed, a direct comparison of results is further complicated. Furthermore, the ranges of variables considered and the accuracy of maintaining control of these variables differed from experiment to experiment.

The experiments reported herein dealing with spot cooling reveal, in general, the same trends.

Arrays of circular jets. - This type of configuration is perhaps the most applicable for cooling turbine components, especially the suction and pressure midchord regions of turbine vanes and blades. Care must be taken to properly space the holes so that jet interference is minimized. Reference 20 reported best heat-transfer results with hole spacings of from 4 to 6 hole diameters (about 5 to 2 percent free-flow area). Results of tests reported in reference 16 showed that the best correlation could be obtained by use of the arrival velocity and the center-to-center hole spacing as the characteristic values in the Reynolds number. This reference also reported that a single jet yielded better heat-transfer results than one jet in an array of jets. Relatively good agreement was found by comparing the results of several different circular array investigations. The effect of crossflow on heat transfer was accounted for in reference 24 by means of a correction parameter obtained from a plot of experimental data; the curve showed that a decrease in heat transfer resulted from increasing the crossflow to the jet. Another investigation (ref. 26) proposed crossflow effects could be accounted for in determining

20

the average Stanton number by including in the correlation a power of the ratio of the crossflow to the jet flow.

Single slot jets. - Data for slot jets impinging on a flat surface were obtained by a number of investigators. Average heat-transfer coefficients obtained by several investigators were compared, and good agreement was found. Reference 6 compared the heat transfer from a slot jet and a circular jet when the same target area, the same flow rate, and the same nozzle area were considered; the round jet yielded heat transfer 8 percent higher than the slot jet. However, the shapes of the areas cooled were quite different, and it was concluded that the shape of the area to be cooled is of primary importance in deciding which type of jet to use. Data obtained for a slot jet could be accurately predicted in the wall jet region; the actual prediction is quite complicated, however.

Arrays of slot jets. --Data for two distinct arrays of slots showed that the identity of each jet was preserved at small slot-to-plate spacings, but jet interaction before impingement occurred at larger spacings. The average data obtained for multiple slots were successfully correlated and, for large values of x_n/B_o, agreed very well with the results of reference 6 for a single slot. Jet interaction caused discrepancies for lower values of x_n/B_o.

When a crossflow equal to as much as 60 percent of the jet flow was superimposed on an array of slot jets, no reduction in heat transfer was found.

Application of Correlations to Turbine Vane Suction and Pressure Surfaces

In order to determine how the correlations for arrays of jets compared, actual dimensions of a turbine vane were inserted into a number of the correlations; and average Nusselt numbers, applicable for the vane suction and pressure surfaces, were calculated. The results were presented in reference 31 and are reproduced herein in figure 19. It should be pointed out that impingement to the suction or pressure surface of a vane is affected by a crossflow set up by the air after it impinges on the surface and flows chordwise to a film cooling hole or a split trailing edge. The effects of a crossflow would tend to reduce the effectiveness of impingement cooling. Figure 19 does not account for any crossflow. However, the correlation presented in reference 24 has in it a term to account for crossflow effects; this term was evaluated for zero crossflow in the preparation of figure 19 so that a fair comparison of the correlations could be made. Suffice it to say that the line representing the correlation of reference 24 would be lowered if crossflow effects were considered. In order to incorporate some kind of safety factor in current designs, it appears that the correlations of reference 16 or 24 should be used until further experiment results in improved correlations.

CONCLUDING REMARKS

From the survey of the heat-transfer characteristics of single and multiple turbulent air jets impinging on flat surfaces, the following conclusions are made:

1. Measured heat-transfer coefficients in the stagnation region exceed those determined by use of the several analytical procedures discussed herein. The analyses are affected by a lack of knowledge of the jet turbulence characteristics. Further investigations leading to more precise information on the jet turbulence characteristics appear to be required.

2. Difficulties in comparing heat-transfer data for single circular jets impinging on flat surfaces were encountered. Consequently, it is difficult to recommend a correlation for use when a single circular jet impinges on a flat surface.

3. Additional investigations of single turbulent room-temperature air jets impinging on heated flat surfaces would be useful.

4. Of primary importance in the design of future turbine vanes and blades is the use of arrays of holes to impingement cool the midchord suction and pressure surfaces; these surfaces can be approximated by flat plates. The correlation of Gardon (ref. 16) appears to be applicable. However, other evidence indicates the hole spacings should be maintained at from 4 to 6 hole diameters to avoid interference between the jets. The use of the arrival velocity in the suggested correlation is attractive. Crossflow effects might be accounted for by use of a crossflow-to-jet-flow parameter, as suggested by Metzger (ref. 26). The correlation of Metzger, where the Reynolds number is based on an equivalent slot width, is also worthy of consideration. Another possibility is to use the correlation of Kercher. It accounts for the effects of crossflow. However, it involves the use of several quantities which must be found by graphical means from a limited amount of experimental data.

5. For single slot jets, Cadek recommends use of the correlation of Gardon (ref. 27) for preliminary design calculations because of its simplicity and ease of calculation.

6. It was found that data for arrays of slots would correlate by the correlation developed by Metzger (ref. 6) for a single slot for large values of slot spacing to slot width. For low values of this parameter, jet interference came into prominence.

7. Wall jet region analyses have been limited to slot jets to date. An analysis for circular jets would probably be in order.

8. For the same flow rate, same nozzle area, and same target area, a circular jet produced heat-transfer coefficients 8 percent higher than those of a slot jet. However, the shape of the area cooled should be considered when determining which type of jet to use.

9. From structural considerations alone, it appears that circular jets are preferable to slot jets for cooling turbine vane and blade surfaces.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, February 5, 1973,
501-24.

APPENDIX - SYMBOLS

A_f open area

B   width of potential core

B_{es} equivalent slot width

B_o slot width

b   half-width of free jet

b'   distance from edge of boundary layer to edge of wall jet

C $\sim D^{3/2}$

C_1 dimensionless potential-core length

c_p specific heat

D   nozzle diameter

d   target diameter or half-length of target

d_h hydraulic diameter

d'   calorimeter diameter

F   function

G   mass flow rate per jet exit area

G'   mass flow rate per unit area of nozzle array

h   heat-transfer coefficient

h_o heat-transfer coefficient at stagnation point

h_r heat-transfer coefficient at radial distance r

K   constant

k   thermal conductivity

L   unheated starting length

l half-length of cooled surface

M* m_c / m_j

m   Euler number, term in correlation obtained graphically, or exponent on Reynolds number

m_c mass flow of crossflow

m_j mass flow of jet

Nu   Nusselt number, hr/k or hd'/k

24

Nu_{B_o}	Nusselt number, hB_o/k
Nu_D	Nusselt number, hD/k
Nu_o	Nusselt number, h_oD/k
n	exponent
Pr	Prandtl number, $c_p\mu/k$
p_s	static pressure
p_t	total pressure
$\dot{q}$	heat flux
$\dot{q}_{lam}$	calculated laminar heat flux
r	distance along plate from stagnation point
Re_a	$\rho U_a D/\mu$ or $\rho U_a x_n/\mu$
Re_{B_o}	$\rho UB_o/\mu$
Re_D	$\rho UD/\mu$
$Re_{D/2}$	$(\rho UD/2)/\mu$
Re_d	$\rho Ud/\mu$
Re_{d_h}	$\rho Ud_h/\mu$
$Re_{d'}$	$\rho Ud'/\mu$
Re_{es}	$\rho U2B_{es}/\mu$
Re_r	$\rho Ur/\mu$
Re_{x_n}	$\rho Ux_n/\mu$
$Re_{y_{1/2}}$	$\rho Uy_{1/2}/\mu$
Re_{2l,U_a}	$\rho U_a 2l/\mu$
St	Stanton number, $Nu/Re\,Pr$
T_o	jet temperature
T_w	wall temperature
U	velocity in axial direction
U_a	arrival velocity
U_m	maximum velocity in axial direction

V	velocity in radial direction
V_m	maximum velocity in radial direction
V_{RB}	reference velocity
x	distance from nozzle in axial direction
x_n	spacing, center to center
Y	spacing, row to row
y	distance from centerline in radial direction
$y_{1/2}$	value of y for $U = U_m/2$
z	distance from impingement plate in axial direction
z_n	nozzle-to-plate distance
$z_{1/2}$	value of z for $V = V_m/2$
δ	boundary-layer thickness
ϵ	degree of turbulence at jet axis
μ	viscosity
ν	kinematic viscosity
ρ	density

Superscript:

| — | average |

REFERENCES

1. Gauntner, James W.; Livingood, John N. B.; and Hrycak, Peter: Survey of Literature on Flow Characteristics of a Single Turbulent Jet Impinging on a Flat Plate. NASA TN D-5652, 1970.

2. Hrycak, Peter; Lee, David T.; Gauntner, James W.; and Livingood, John N. B.: Experimental Flow Characteristics of a Single Turbulent Jet Impinging on a Flat Plate. NASA TN D-5690, 1970.

3. Bradbury, L. J. S.: Simple Expressions for the Spread of Turbulent Jets. Aeron. Quart., vol. 18, pt. 2, May 1967, pp. 133-142.

4. Kezios, Stothe P.: Heat Transfer in the Flow of a Cylindrical Air Jet Normal to an Infinite Plane. Ph.D. Thesis, Illinois Inst. Tech., 1956.

5. Walz, Dieter R.: Spot Cooling and Heating of Surfaces with High Velocity Impinging Air Jets. Part 2 - Circular Jets on Plane and Curved Surfaces, Slot Jets on Curved Surfaces. TR No. 61, Stanford Univ. (AD-607727), July 1964.

6. Metzger, Darryl E.: Spot Cooling and Heating of Surfaces with High Velocity Impinging Air Jets. Part 1 - Slot Jets on Plane Surfaces. TR No. 52, Stanford Univ., Apr. 1962.

7. Schauer, J. J.; and Eustis, R. H.: The Flow Development and Heat Transfer Characteristics of Plane Turbulent Impinging Jets. Tech. Rep. 3, Dept. Mech. Eng., Stanford Univ. Sept. 30, 1963.

8. Myers, G. E.; Schauer, J. J.; and Eustis, R. H.: Heat Transfer to Plane Turbulent Wall Jets. J. Heat Transfer, vol. 85, no. 3, Aug. 1963, pp. 209-214.

9. Cadek, Frederick F.: A Fundamental Investigation of Jet Impingement Heat Transfer. Ph.D. Thesis, Univ. Cincinnati, 1968.

10. Eckert, E. R. G.; and Livingood, John N. B.: Method for Calculation of Laminar Heat Transfer in Air Flow Around Cylinders of Arbitrary Cross Section. NACA Rep. 1118, 1953.

11. Tomich, John F.: Heat and Momentum Transfer from Compressible Turbulent Jets of Hot Air Impinging Normally on a Surface. Ph.D. Thesis, Washington Univ., 1967.

12. Brdlik, P. M.; and Savin, V. K.: Heat Transfer Between an Axisymmetric Jet and a Plate Normal to the Flow. J. Eng. Phys., vol. 8, Feb. 1965, pp. 91-98.

13. Perry, K. P.: Heat Transfer by Convection from a Hot Gas Jet to a Plane Surface. Proc. Inst. Mech. Eng., vol. 168, no. 30, 1954, pp. 775-780.

14. Thurlow, G. G.: Communications on Heat Transfer by Convection. Proc. Inst. Mech. Eng., vol. 168, no. 30, 1954, pp. 781-783.

15. Huang, G. C.: Investigations of Heat-Transfer Coefficients for Air Flow Through Round Jets Impinging Normal to a Heat-Transfer Surface. J. Heat Transfer, vol. 85, no. 3, Aug. 1963, pp. 237-245.

16. Gardon, Robert; and Cobonpue, John: Heat Transfer Between a Flat Plate and Jets of Air Impinging on It. International Developments in Heat Transfer. ASME, 1963, pp. 454-460.

17. Chamberlain, John E.: Heat Transfer Between a Turbulent Round Jet and a Segmented Plate Perpendicular to It. M.S. Thesis, Newark College of Engineering, 1966.

18. Donaldson, Coleman DuP.; Snedeker, Richard S.; and Margolis, David P.: A Study of Free Jet Impingement. Part 2. Free Jet Turbulent Structure and Impingement Heat Transfer. J. Fluid Mech., vol. 45, pt. 3, Feb. 15, 1971, pp. 477-512.

19. Rosenbaum, H.; and Donaldson, C. DuP.: An Analysis of Jet Impingement Heat Transfer. Rep. 101, Aero. Res. Associates of Princeton, Inc., Mar. 1967.

20. Freidman, S. J.; and Mueller, A. C.: Heat Transfer to Flat Surfaces. Proceedings of the General Discussion on Heat Transfer. Inst. Mech. Eng., 1951, pp. 138-142.

21. Ott, Hanns H.: Heat Transfer to a Plate Cooled by Air Flow. Schweiz. Bauzeitung, vol. 79, no. 46, Nov. 16, 1961, pp. 834-840.

22. Hilgeroth, E.: Heat Transfer in Jet Flow Perpendicular to the Transfer Surface. Chem.-Ing.-Tech., vol. 37, no. 12, Dec. 1965, pp. 1264-1272.

23. Kercher, David: Heat Transfer by a Multiple Array of Round Air Jets Impinging Perpendicular to a Flat Surface Including Effects of Spent Air. M.S. Thesis, Univ. of Cincinnati, 1967.

24. Kercher, D. M.; and Tabakoff, W.: Heat Transfer by a Square Array of Round Air Jets Impinging Perpendicular to a Flat Surface Including the Effect of Spent Air. J. Eng. Power, vol. 92, no. 1, Jan. 1970, pp. 73-82.

25. Burggraf, Frederick: Average Heat Transfer Coefficients with a Row of Air Jets Discharging into a Half Cylinder. M.S. Thesis, Univ. Cincinnati, 1967.

26. Metzger, D. E.; and Korstad, R. J.: Effects of Crossflow on Impingement Heat Transfer. Paper 71-GT-1, ASME, Mar. 1971.

27. Gardon, Robert; and Akfirat, J. Cahit: Heat Transfer Characteristics of Impinging Two-Dimensional Air Jets. J. Heat Transfer, vol. 88, no. 1, Feb. 1966, pp. 101-108.

28. Cartwright, W. G.; and Russell, P. J.: Characteristics of a Turbulent Slot Jet Impinging on a Plane Surface. Proc. Inst. Mech. Eng., vol. 182, pt. 3H, 1967-68, pp. 309-319.

29. Andreyev, A. A.; Dakhno, V. N.; Savin, V. K.; and Yadayev, B. N.: Investigation of Heat Transfer in a Gradient Flow Region for Plane Turbulent Jet Impinging on a Plate Situated Normal to the Flow. NASA TT F-14195, 1972.

30. Schuh, H.; and Pettersson, R.: Heat Transfer by Arrays of Two-Dimensional Jets Directed Normal to Surfaces Including the Effects of a Superposed Wall-Parallel Flow. Proceedings of the Third International Heat Transfer Conference. Vol. 2. AIChE, 1966, pp. 280-291.

31. Livingood, John N. B.; Ellerbrock, Herman H.; and Kaufman, Albert: 1971 NASA Turbine Cooling Research Status Report. NASA TM X-2384, 1971.
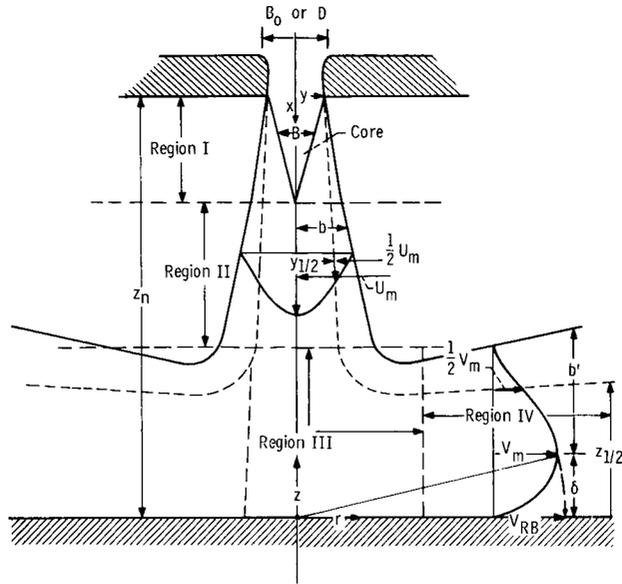
(a) Characteristic regions in impinging jet flow. (From ref. 1.)
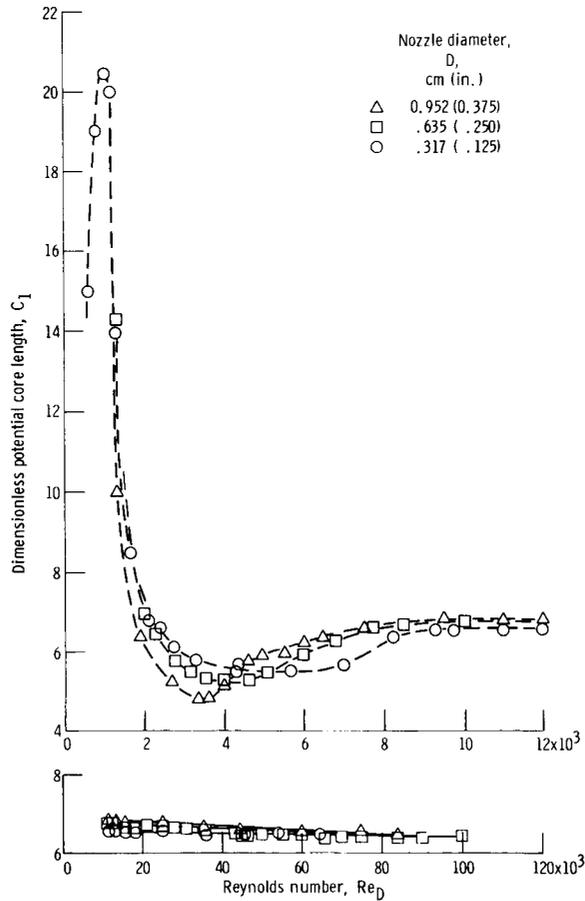
(b) Influence of nozzle diameter on impinging jet flow patterns. (From ref. 2.)

Figure 1. – Jet flow characteristics.

30
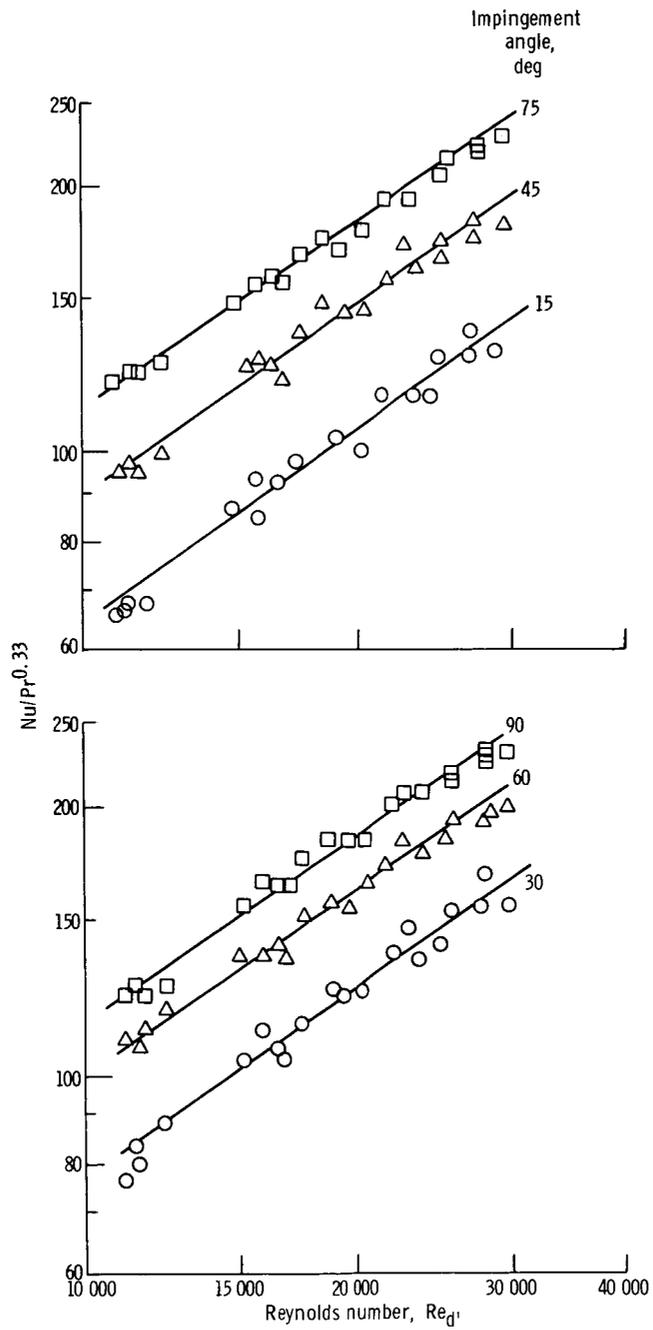
Figure 2. - Effect of impingement angle on heat transfer. (From ref. 13.)
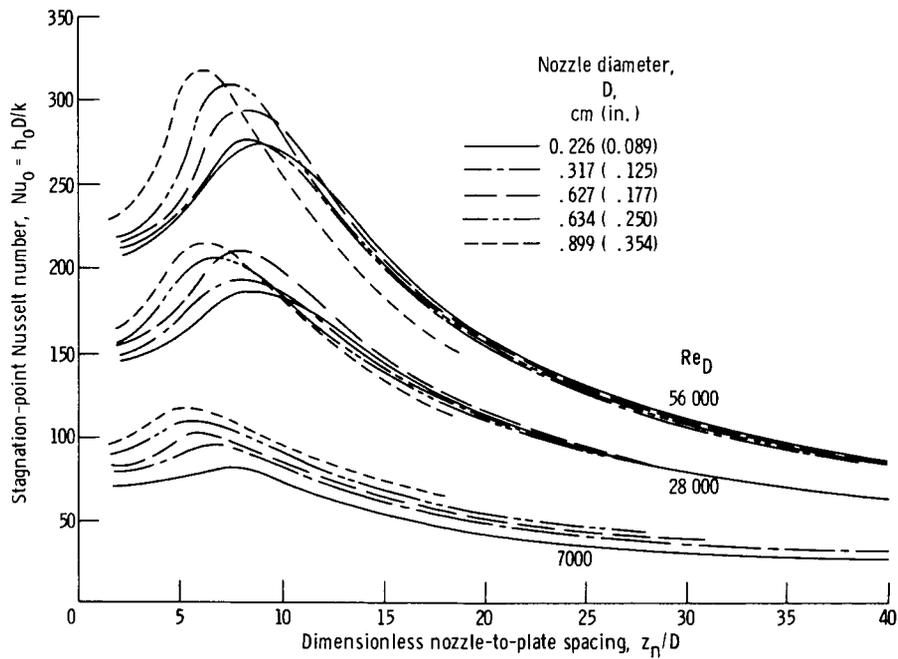
Figure 3. - Correlation of heat-transfer coefficient at stagnation point of a jet. (From ref. 16.)
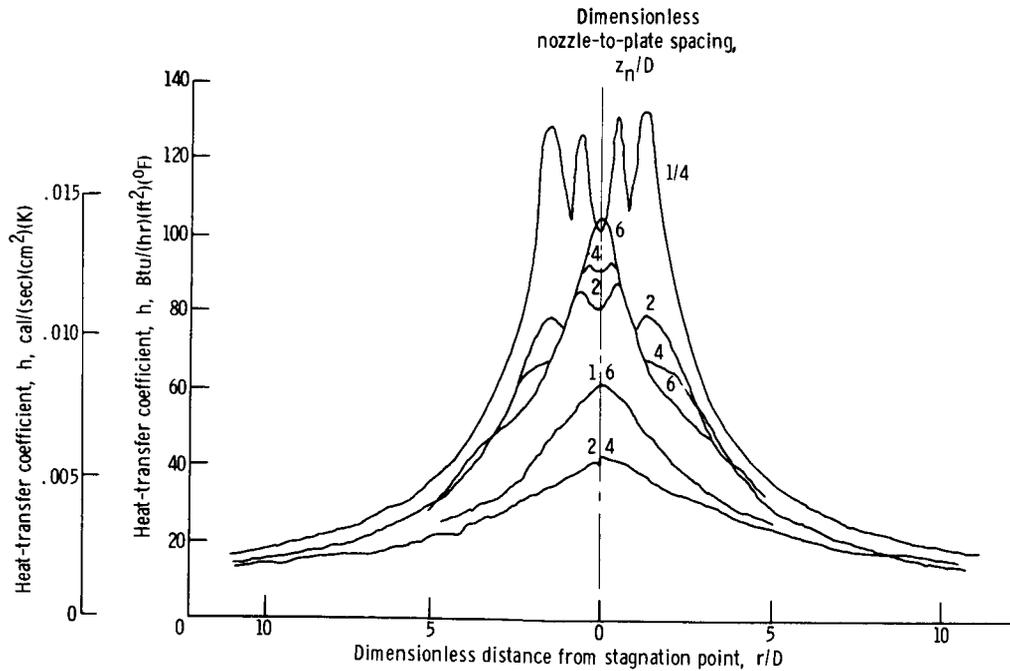
Figure 4. - Radial variation of heat transfer between a plate and an impinging jet. Nozzle diameter D = 0.634 centimeter (0.250 in.); Re_D = 28 000; temperature difference ΔT = 20 K (36° F). (From ref. 16.)
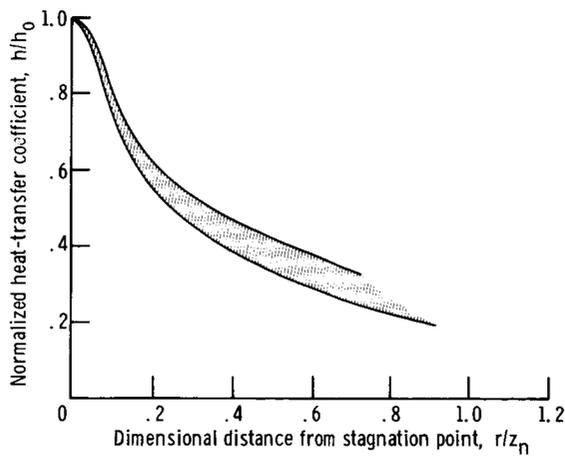
Figure 5. – Radial variation of normalized heat-transfer coefficients between a plate and an impinging jet. Dimensionless nozzle-to-plate spacing $z_n/D > 10$; $Re_D > 7000$. (From ref. 16.)
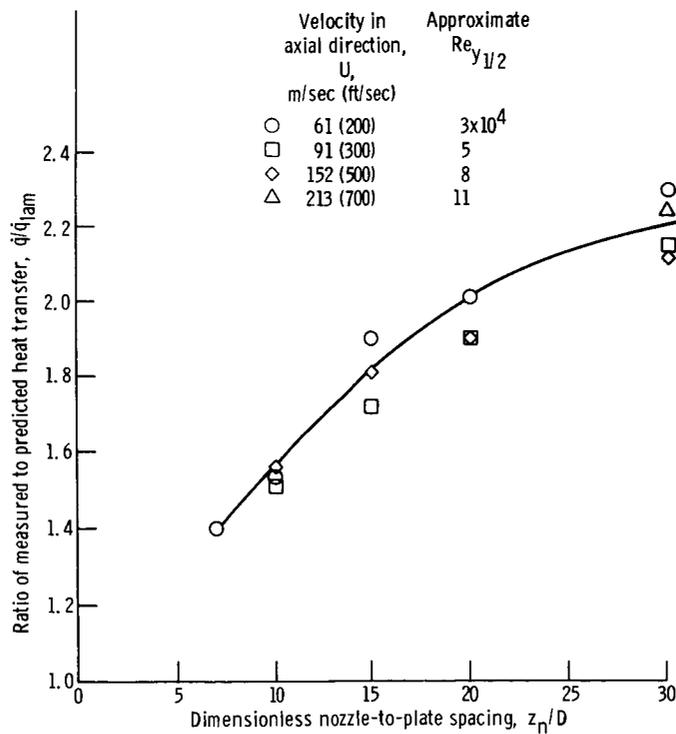
Figure 6. – Ratio of measured heat transfer to that predicted by laminar theory at stagnation point of impinging jet. (From ref. 18.)
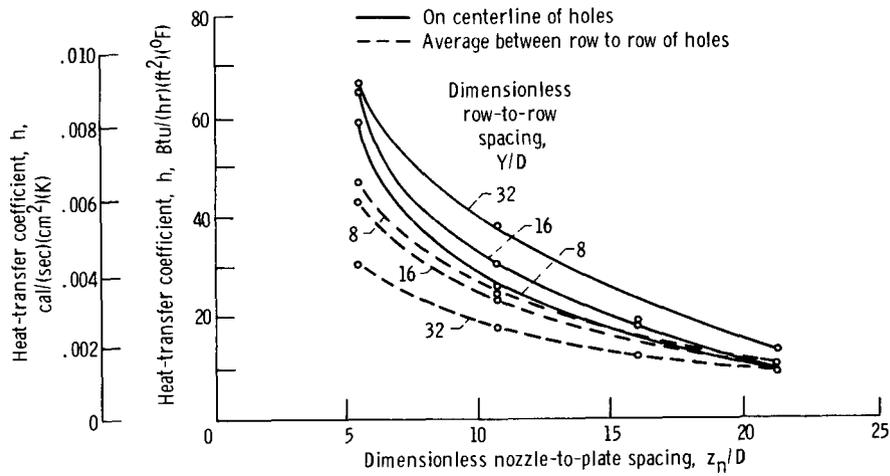
Figure 7. - Variation of heat-transfer coefficient under simple multiple jets with axial distance ratio z_n/D. Nozzle diameter $D = 0.476$ centimeter (3/16 in.); center-to-center spacing $x_n = 1.588$ centimeters (5/8 in.); row-to-row spacing $Y = 3.81, 7.62, 15.24$ centimeters (1.5, 3, 6 in.). (From ref. 15.)
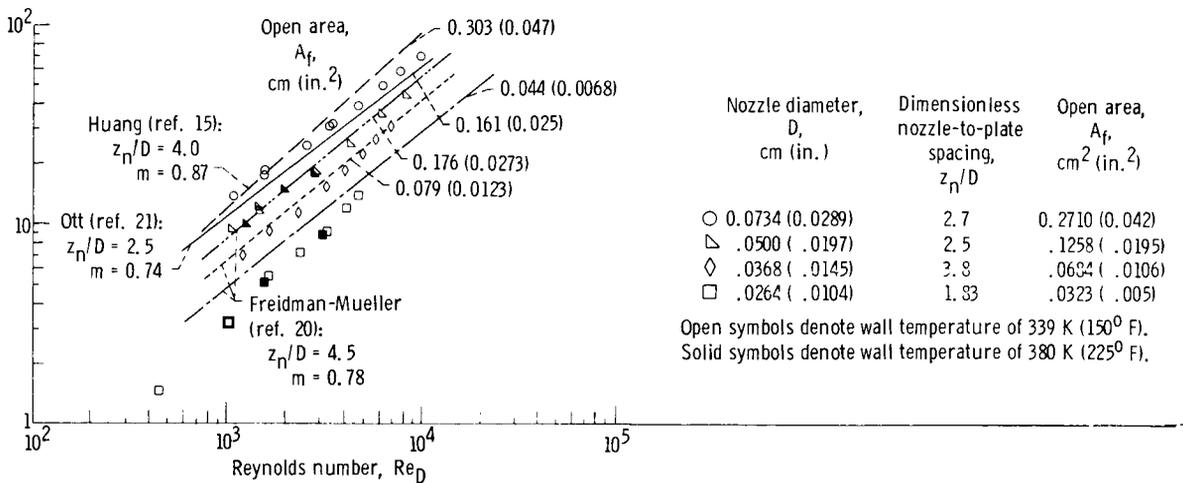
(a) Test results of reference 23 compared to results of references 15, 20, and 21. Constant hole spacing, $x_n = 0.318$ centimeter (0.125 in.); jet temperature, 297 K (75° F).
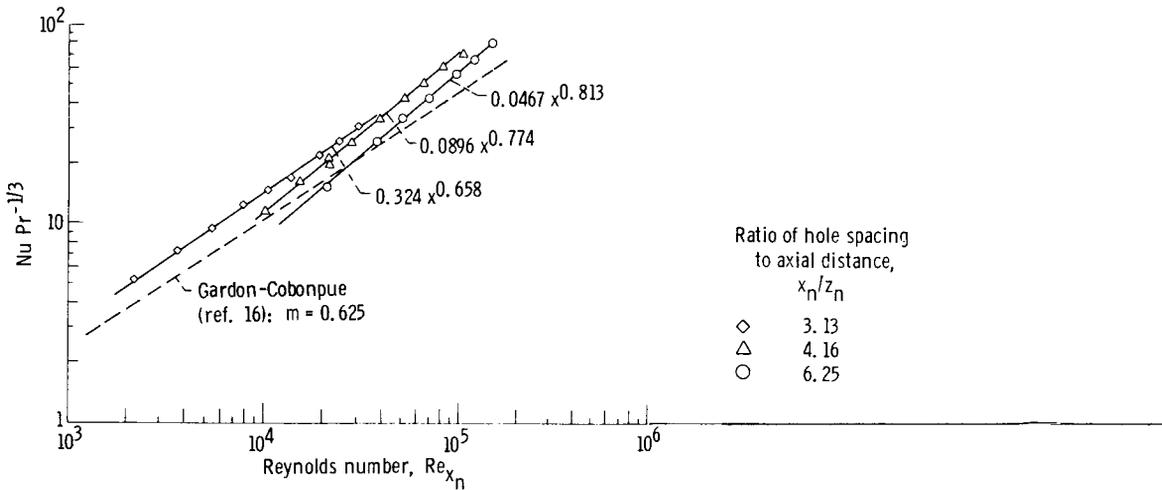
(b) Test results of reference 23 compared to results of reference 16. Wall temperature $T_w = 339$ K (150° F).
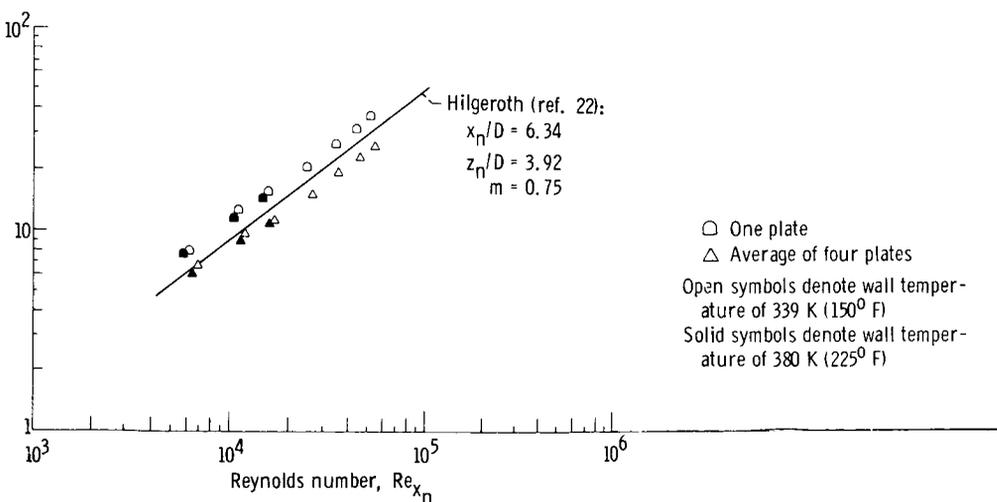
(c) Test results of reference 23 compared to results of reference 22.

Figure 8. – Heat transfer of multiple array of impinging air jets. (From ref. 23.)
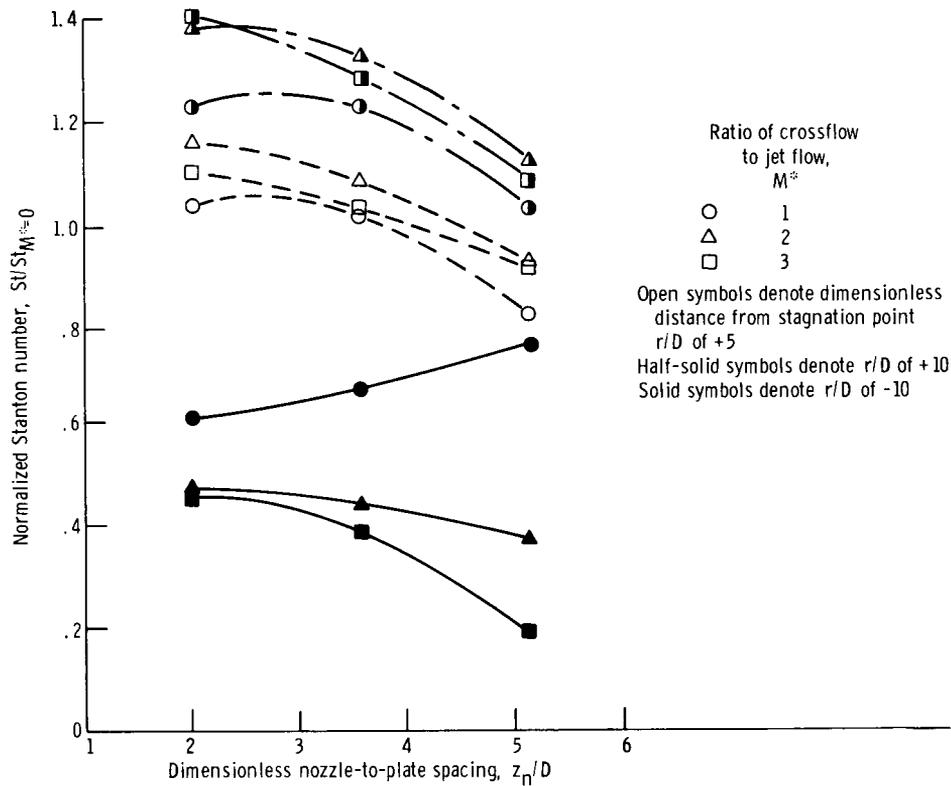
Figure 9. – Effect of nozzle-to-plate spacing on normalized Stanton number for row of holes with center-to-center hole spacing of three hole diameters. Dimensionless hole spacing $x_n/D = 3.0$; equivalent-slot Reynolds number $Re_{es} = 2000$. (From ref. 26.)
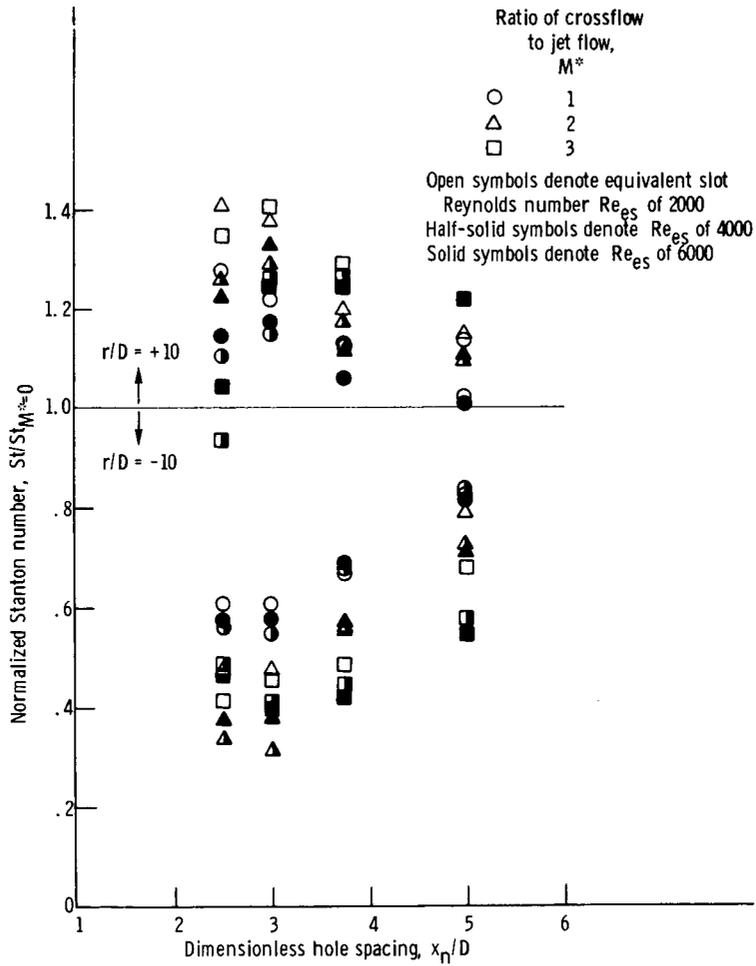
Figure 10. – Effect of center-to-center hole spacing on normalized Stanton number for a row of holes with nozzle-to-plate spacing of 2 nozzle diameters. (From ref. 26.)
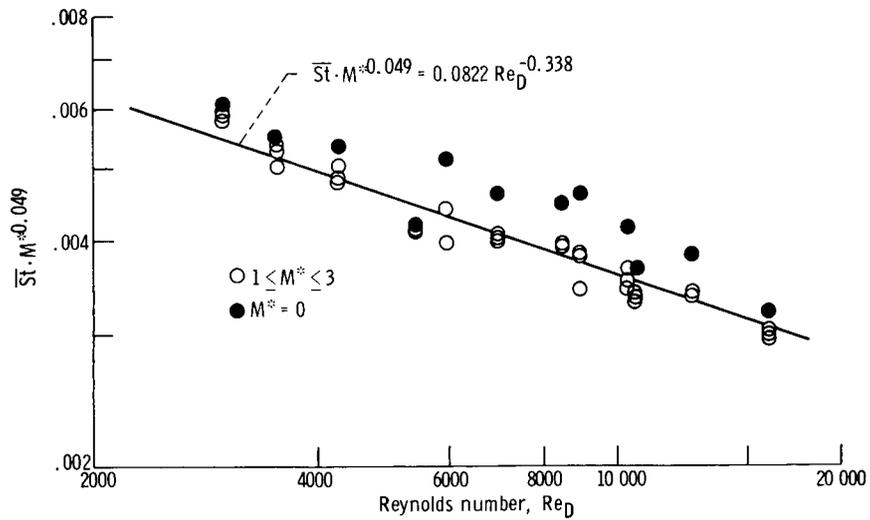
Figure 11. - Empirical correlation of the overall effect of crossflow for a row of holes. Dimensionless nozzle-to-plate spacing $z_n/D = 2$; ratio of unheated starting length to nozzle diameter $L/D = \pm 10$; dimensionless hole spacing, $2.5 \leq x_n/D = 5.0$. (From ref. 26.)
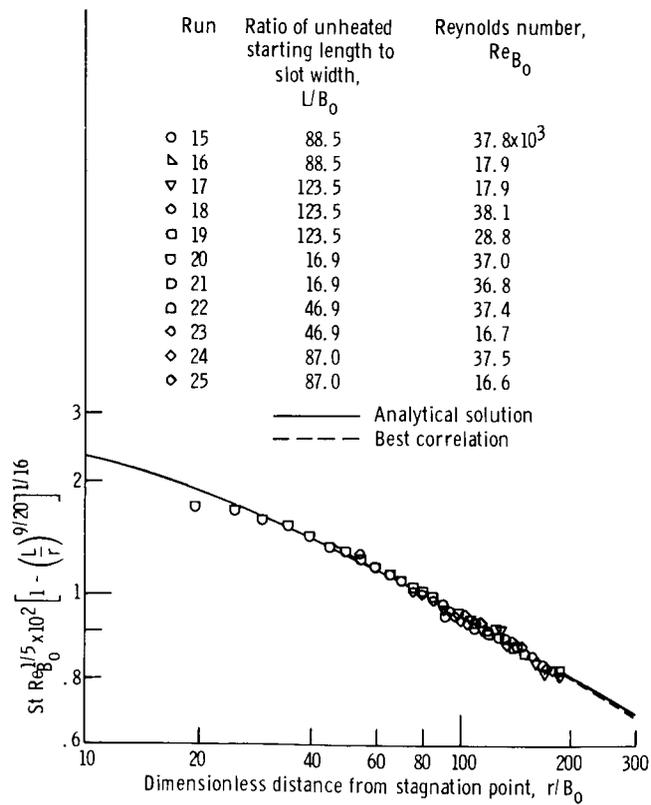
Figure 12. - Heat transfer to wall jets from an isothermal wall with an unheated starting length (correlated data). (From ref. 8.)

Slot-width   Slot width,
Reynolds number, B_o,
Re_{B_0} cm (in.)

○   4 650   0.635 (0.25)
□   11 400
◇   20 750
◺   50 300
◠   76 500

Open symbols denote dimensionless
nozzle-to-plate spacing z_n/B_0 of 8
Solid symbols denote z_n/B_0 of 16

Correlation of reference 27 ($8 \leq z_n/B_0 \leq 64$):

$$\overline{Nu}_{2l} = 0.36\left(Re_{2l, U_a}\right)^{0.62}$$

(Solid line indicates range of data)

Average Nusselt number, $\overline{Nu}_{2l} = \overline{h}2l/k$

Reynolds number, $Re_{2l, U_a} = \dfrac{U_a 2l}{\nu}$
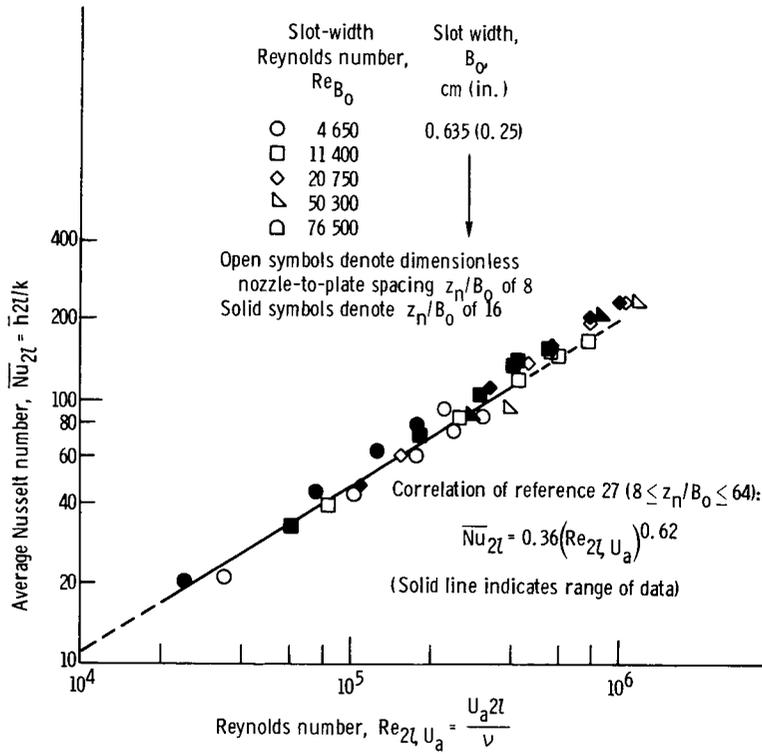
Figure 16. – Comparison of average heat-transfer data for a slot jet
with empirical correlation of reference 27. (From ref. 9.)
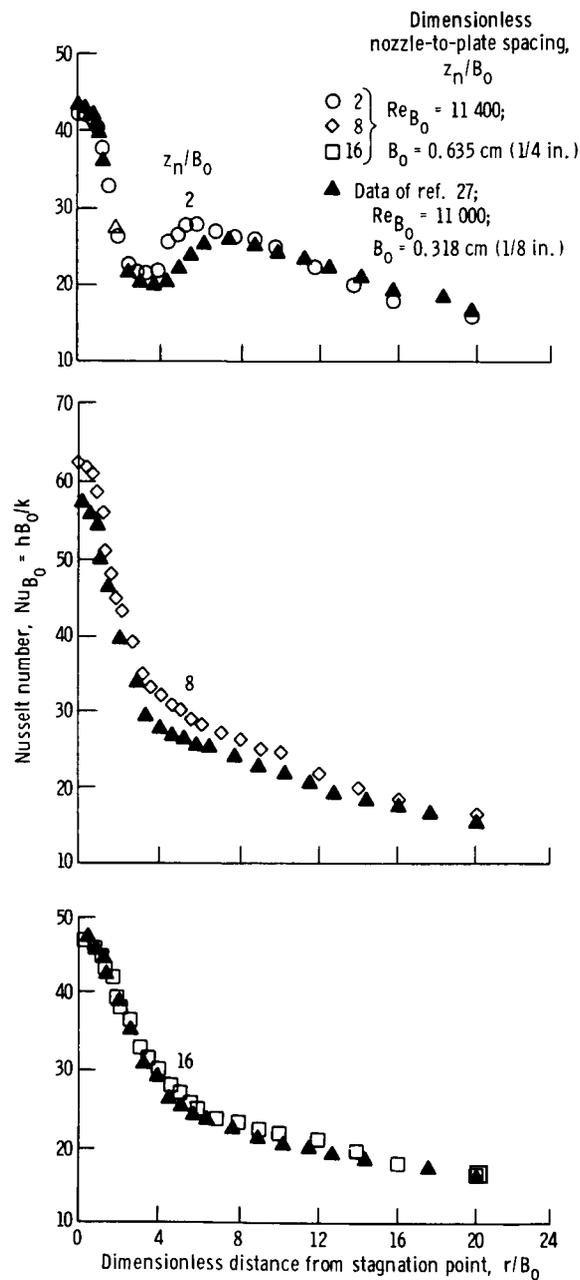
Figure 14. – Comparison of local Nusselt number measurements for a slot jet with those of reference 27. (From ref. 9.)

Slot-width
Reynolds number,
Re_{B_0}

Slot width,
B_0
cm (in.)

Dimensionless
nozzle-to-plate
spacing,
z_n/B_0

	Re_{B_0}	B_0 cm (in.)	z_n/B_0
○	4 650	0.635 (0.25)	8
□	11 400	.635 (.25)	8
◇	20 750	.635 (.25)	8

—— Correlation of reference 6 ($2980 \le Re_{B_0} \le 7400$; $7 \le z_n/B_0 \le 10$):

$$\frac{\overline{Nu}_{B_0}}{\left(Re_{B_0}\right)^{0.566}} = 0.656 \left(\frac{l}{B_0}\right)^{-0.434}$$

$\dfrac{\overline{Nu}_{B_0}}{\left(Re_{B_0}\right)^{0.566}}$

Dimensionless half-length of cooled surface, l/B_0
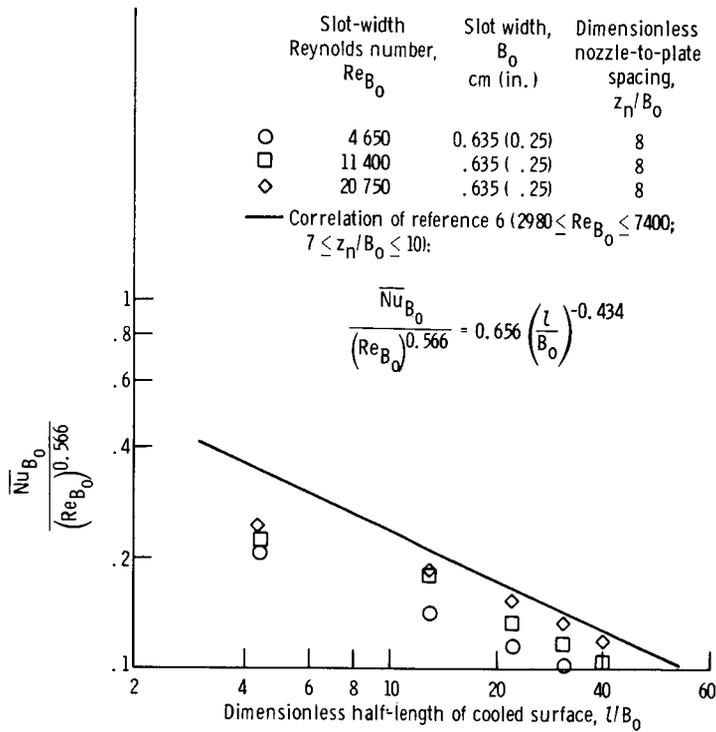
Figure 15. - Comparison of average heat-transfer data for a slot jet with empirical correlation of reference 6. (From ref. 9.)

Slot width,
B_0,
cm (in.)

	B_0 cm (in.)
□	0.159 (1/16)
△	.318 (1/8)
○	.635 (1/4)
◈	Metzger's data

Open symbols denote center-to-center
spacing x_n of 5.08 cm (2 in.)
Solid symbols denote x_n of
10.16 cm (4 in.)

$\overline{Nu} = Re_a^{0.62}$

Average Nusselt number, $\overline{Nu} = \bar{h}x_n/k$

Reynolds number, $Re_a = \dfrac{U_a x_n \rho}{\mu} = 2.65 \dfrac{Gx_n^2}{\mu\sqrt{B_0 z_n}}$
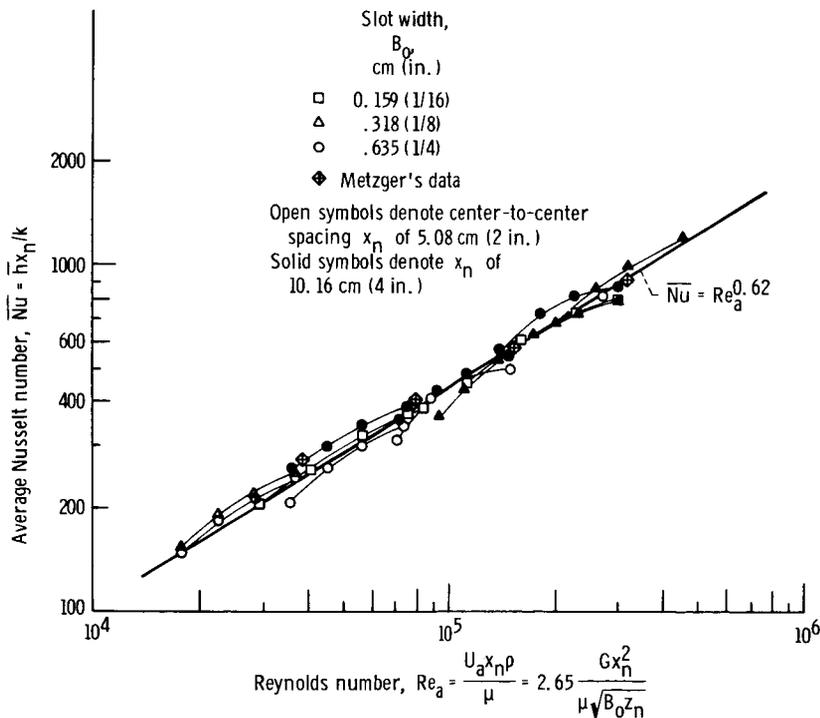
Figure 13. - Correlation of average heat-transfer coefficients between a target of finite width and a single two-dimensional air jet. Mass flow rate per jet exit area G = 3515 to 28 123 kg/(hr)(m²), or 720 to 5760 lb/(hr)(ft²); dimensionless nozzle-to-plate spacing z_n/B_0 = 8 to 64. (From ref. 27.)
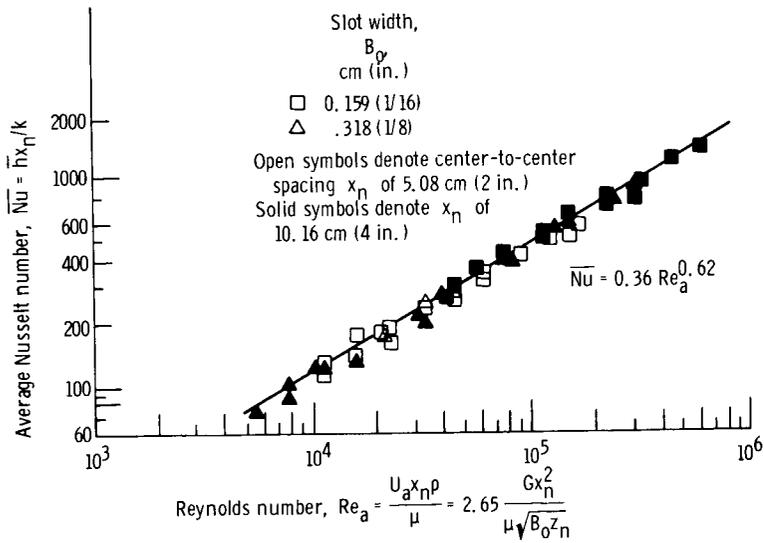
Figure 17. – Correlation of average heat-transfer coefficients between a plate and arrays of impinging two-dimensional jets. Mass flow rate per jet exit area G = 977 to 7812 kg/(hr)(m²), or 200 to 1600 lb/(hr)(ft²); dimensionless nozzle-to-plate spacing z_n/B_0 = 8 to 64. (From ref. 27.)
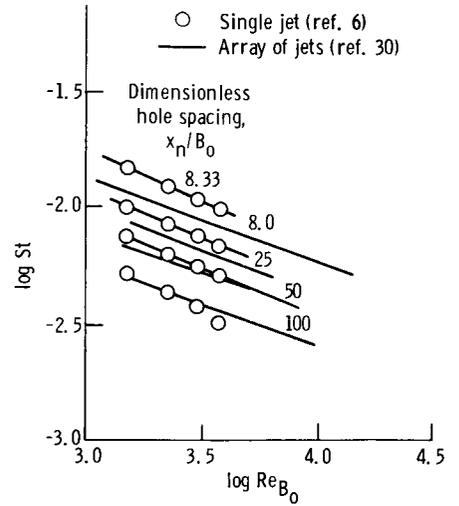
Figure 18. – Single-slot-jet results (ref. 6) compared with results for arrays of slot jets (ref. 30). Dimensionless nozzle-to-plate spacing z_n/B_0 = 8. (From ref. 30.)
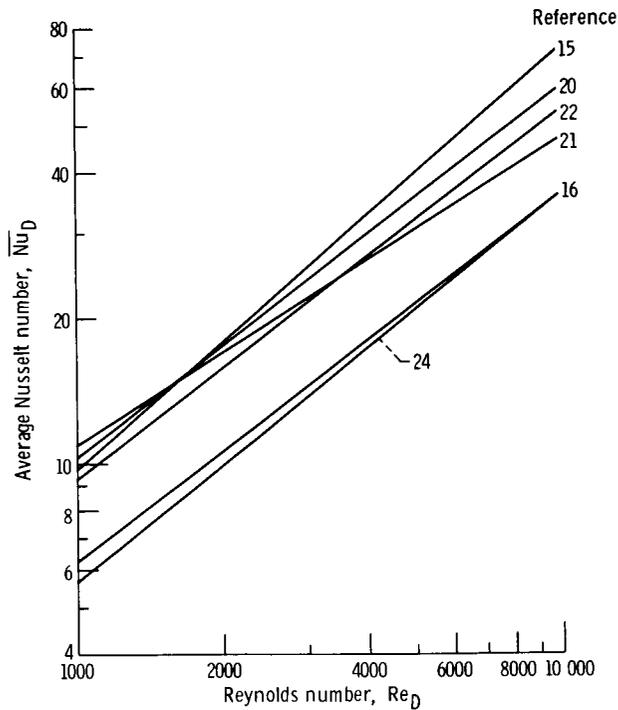
Figure 19. – Nusselt number as function of Reynolds number for two-dimensional array of circular jets as calculated from references 15, 16, 20, 21, 22, and 24. (From ref. 31.)